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(71) Applicant(s)  
New Holland Belgium N.V.  
(Incorporated in Belgium)  
Leon Claeyssstraat 3A, B-8210 Zedelgem,  
Belgium

(72) Inventor(s)  
Antoon Vermeulen

(74) Agent and/or Address for Service  
Alec Moses Messulam  
A Messulam & Co Limited,  
43-45 High Road, BUSHEY HEATH, Herts,  
WD2 1EE, United Kingdom

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(54) Abstract Title  
Hydro-mechanical transmission

(57) An input shaft 18 drives a hydrostatic pump 10 which in turn drives a hydrostatic motor 14. The motor 14 turns an auxiliary shaft 16 and by adjusting swash plates in the pump 10 or motor 14, the transmission ratio between the auxiliary shaft 16 and the input 18 can be varied. The input shaft 18 and auxiliary shaft 16 also connect to a gearbox 20 which has an output shaft 22 which connects to a vehicles wheels. The gearbox 20 has two planetary gear sets 30,40, formed of sun 32,42, planet 34,44 and ring gears 38,48. Both ring gears 38,48 are permanently connected to the output shaft 22. A coupling sleeve 35 can couple the input shaft 18 to one or both gear sets 30,40, another sleeve 45 can couple the auxiliary shaft 16 to one or both gear sets 30,40. A position exists allowing both sleeves 35, 45, and therefore both drive shafts 16, 18 to be connected to the output shaft 22, which allows a power shift when changing gear sets 30,40 but normally only one connection is made.

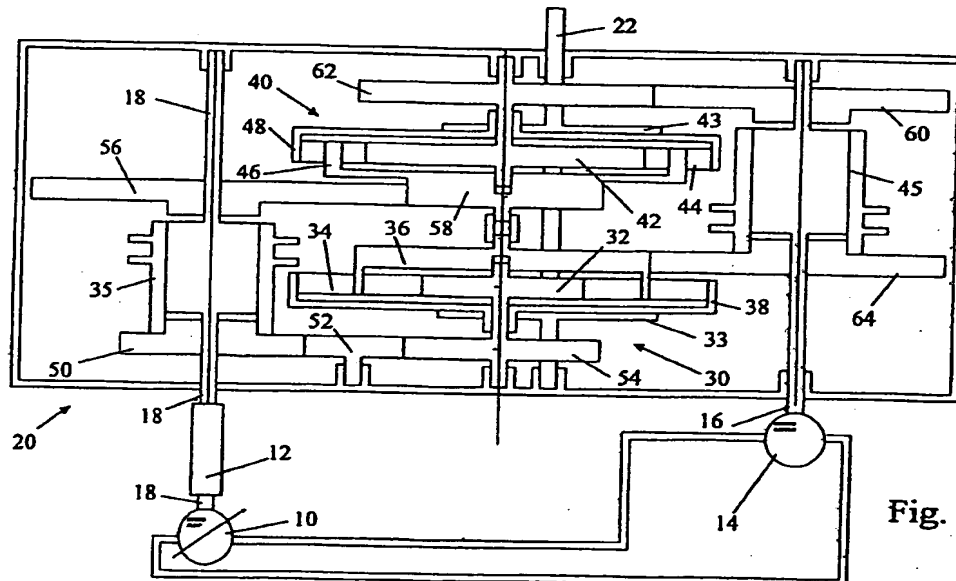


Fig. 1

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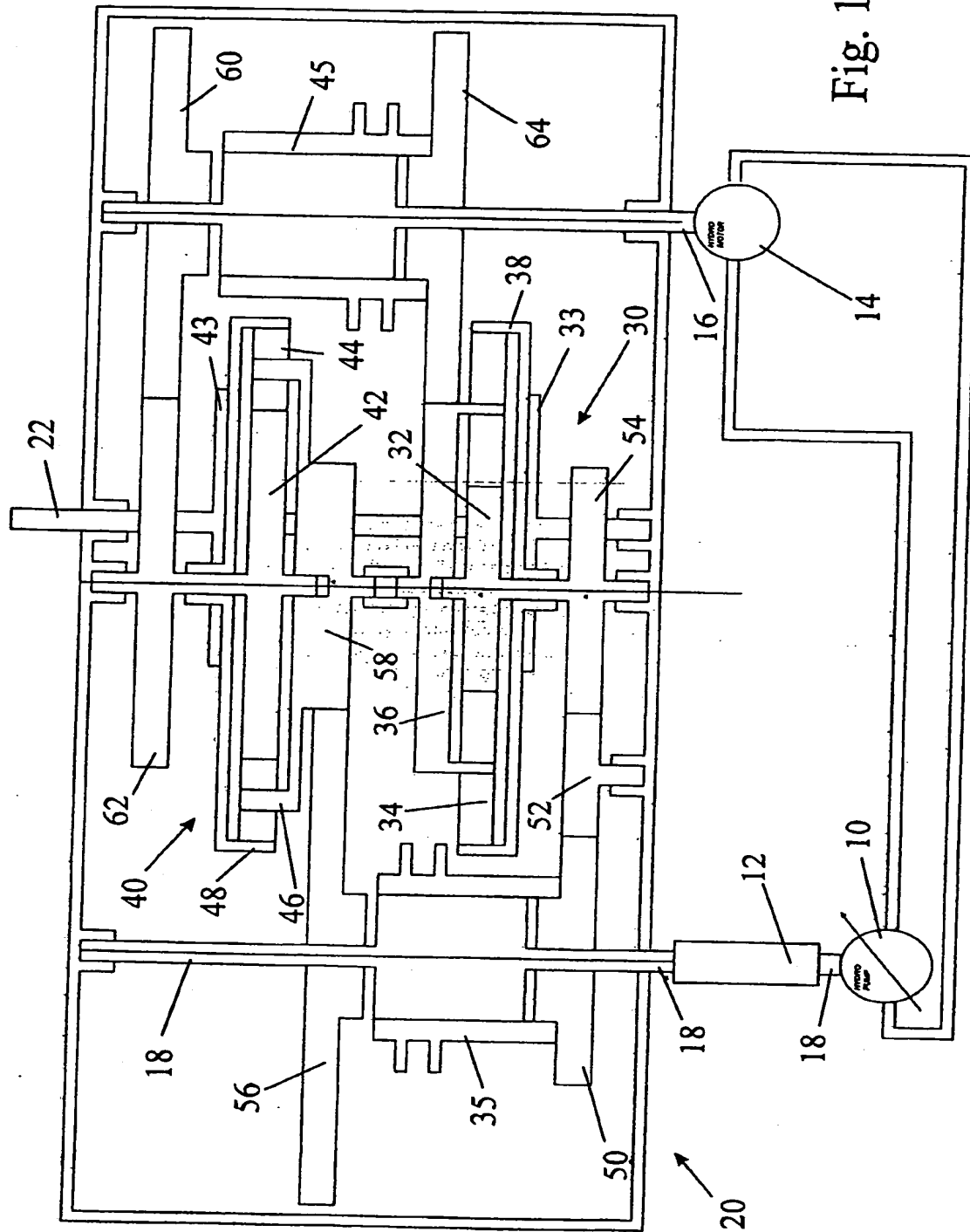


Fig. 1

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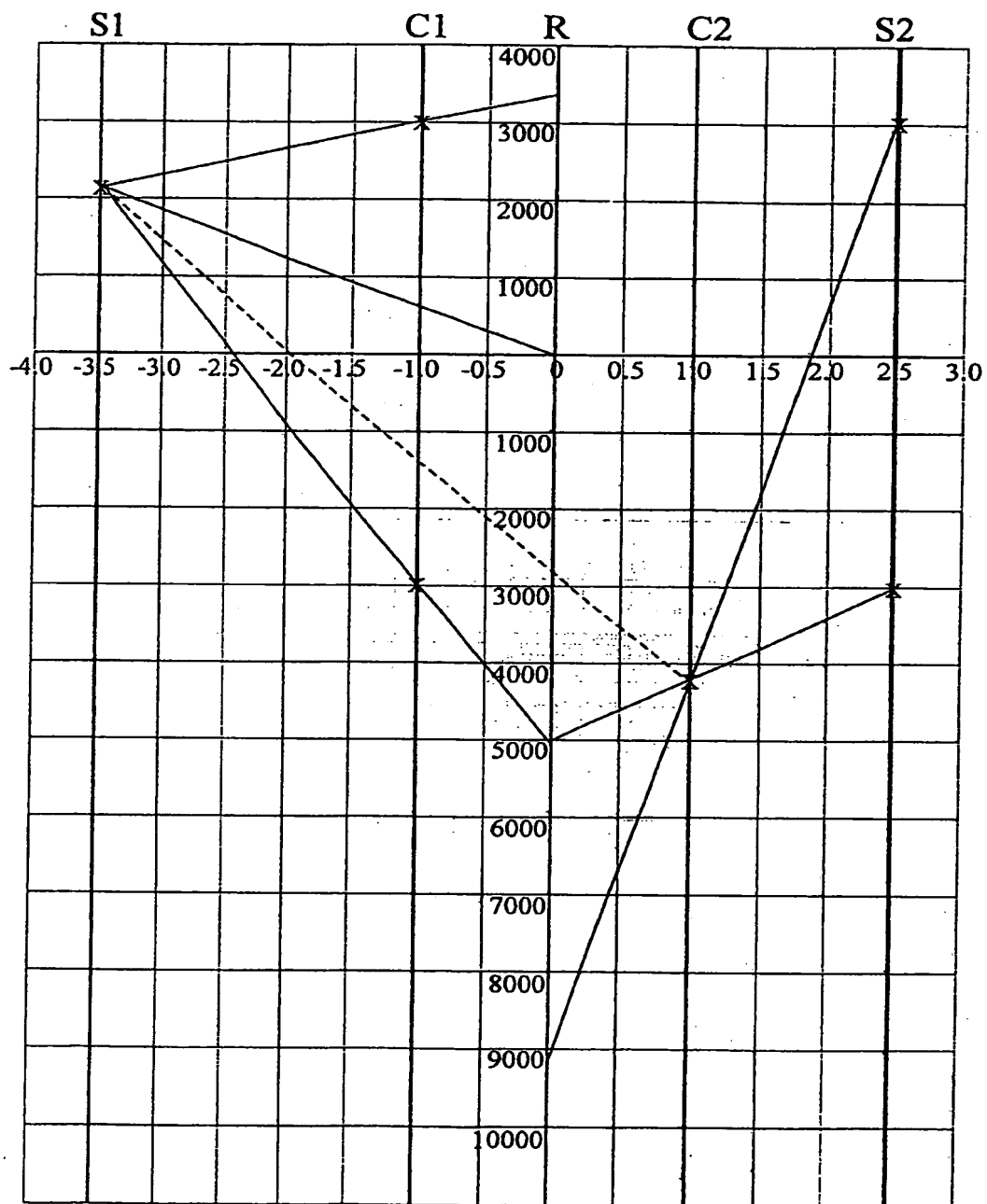


Fig. 2

## TRANSMISSION

The present invention relates to an infinitely variable transmission that employs mechanical and hydraulic or  
5 electrical transmission elements.

Continuously variable gearboxes are known that rely on the use of a variable hydrostatic transmission. Such a transmission may typically provide a transmission ratio that  
10 can be varied continuously from -1.5:1 to +1.5:1. This is considerably less than the range of transmission ratios required between the engine and the driving wheels of a vehicle and of itself a hydrostatic transmission cannot be used as a vehicle gearbox.

15 It has also been proposed in the prior art to use a hydrostatic transmission in combination with a mechanical transmission to extend the range of gear ratios that can be achieved. The prior art gearbox that most closely resembles  
20 the present invention is described in US-A-5,403,241. That gearbox includes a multi-shaft toothed-wheel planetary gearing and a continuously adjustable hydrostatic transmission having toothed-wheel auxiliary transmission stages and gear-changing clutches. The hydrostatic  
25 transmission is associated with a displacement-type machine with constant displacement volume. In the planetary gearing, one shaft forms the input shaft to which the adjustable displacement-type machine is connected. A second shaft of the planetary gearing is connected to the constant-volume  
30 displacement-type machine. Third and fourth shafts of the planetary gearing represent coupling shafts. The coupling shafts alternately act via at least two auxiliary transmission stages, also called gears, on the output shaft. The coupling shafts change their speeds in such a manner  
35 that one coupling shaft becomes steadily faster and the other one becomes steadily slower. They behave in such a manner that they exhibit equal speeds in one extreme

position. In the extreme positions of the hydrostatic transmission, gear-changing clutches handle the gear change free of load and without interruption of traction force when the speeds are synchronous. The automatic transmission is  
5 controlled and regulated by electronic and hydraulic means.

The present invention in its first aspect seeks to provide a continuously variable hydro-mechanical transmission of simpler construction that provides a wide  
10 range of transmission ratios without interruption of power over the entire range.

According to the first aspect of the present invention, there is provided a continuously variable transmission  
15 comprising

an input shaft,  
a hydrostatic pump driven by the input shaft,  
a hydrostatic motor hydraulically connected to the hydrostatic pump and serving to drive an auxiliary shaft  
20 with a continuously variable transmission ratio relative to the input shaft, and

mechanical gearing for transmitting torque from the input and auxiliary shafts to an output shaft,

characterised in that the mechanical gearing comprises  
25 two planetary gear sets arranged to provide parallel torque paths from the input and auxiliary shafts to the output shaft and coupling means for selectively engaging the planetary gear sets,

the first planetary gear set, when engaged by the  
30 coupling means to transmit torque, comprising first a sun gear connected to the input shaft, a first planetary gear carrier connected to the auxiliary shaft and a first ring gear connected to drive the output shaft,

the second planetary gear set, when engaged by the  
35 coupling means to transmit torque, comprising a second sun gear connected to the auxiliary shaft, a second planetary

gear carrier connected to the input shaft and a second ring gear connected to drive the output shaft, and

the coupling means serving to engage the first planetary gear set for reverse and lower transmission ratios, to engage the second planetary gear set for higher transmission ratios and to engage both planetary gear sets simultaneously when the ratio of the speed of rotation of the auxiliary shaft relative to the input shaft is such that both planetary gear sets drive the output shaft at the same speed.

Preferably, the transmission ratios of the planetary gear sets are selected such that the planetary gear sets drive the output shaft at the same speed only at the limit of adjustment of the speed of the auxiliary shaft relative to the input shaft.

There have also been proposed in the past transmissions for hybrid electrical vehicles. Such vehicles have an electric motor drive in addition to an internal combustion engine. The engine drives a generator for charging the batteries and is in addition capable of driving the vehicle wheels directly.

The present invention in a second aspect seeks to provide a continuously variable transmission that is suitable for use in a hybrid vehicle.

In accordance with the second aspect of the invention, there is provided a continuously variable transmission of a hybrid electrical vehicle comprising:

an input shaft connected to an internal combustion engine,

an auxiliary shaft driven by an electric motor deriving power from a bank of batteries charged by the internal combustion engine, the auxiliary shaft being rotatable by the electric motor with a speed that is

continuously variable relative to the speed of rotation of the input shaft, and

mechanical gearing for transmitting torque from the input and auxiliary shafts to an output shaft,

5        characterised in that the mechanical gearing comprises two planetary gear sets arranged to provide parallel torque paths from the input and auxiliary shafts to the output shaft and coupling means for selectively engaging the planetary gear sets,

10       the first planetary gear set, when engaged by the coupling means to transmit torque, comprising first a sun gear connected to the input shaft, a first planetary gear carrier connected to the auxiliary shaft and a first ring gear connected to drive the output shaft,

15       the second planetary gear set, when engaged by the coupling means to transmit torque, comprising a second sun gear connected to the auxiliary shaft, a second planetary gear carrier connected to the input shaft and a second ring gear connected to drive the output shaft, and

20       the coupling means serving to engage only the first planetary gear set for reverse and lower transmission ratios, to engage only the second planetary gear set for higher transmission ratios and to engage both planetary gear sets simultaneously when the ratio of the speed of rotation  
25       of the auxiliary shaft relative to the input shaft is such that both planetary gear sets drive the output shaft at the same speed.

It will be appreciated that the electrical and  
30       hydraulic implementations of the invention are analogous to one another in that in both cases the mechanical gearing is called upon to combine the torque from an input shaft, which may rotate with constant speed, with the torque of an  
35       auxiliary shaft that rotates with variable speed in either direction relative to the input shaft. The only difference is that in one case the input shaft drives the auxiliary shaft hydraulically and in the other case electrically.

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a schematic representation of a hydro-  
5 mechanical transmission of the invention, and

Figure 2 is a graph that shows the speeds of rotation of the different components of the planetary gear sets.

Figure 1 shows a hydro-mechanical transmission of the  
10 invention which is intended for use in an agricultural vehicle, such as a tractor. A hydrostatic pump 10 is driven at constant speed by an input shaft 18 constituted by the crankshaft of a diesel engine 12 and in turn drives a hydrostatic motor 14. The motor drives an auxiliary shaft 16  
15 and by adjustment of swash plates of the hydrostatic pump and/or motor the transmission ratio between the auxiliary shaft 16 and the engine crankshaft can be varied. It will be assumed herein that the engine is operated at a constant speed of 2,100 rpm and that, by suitable setting of the  
20 swash plates, the speed of the auxiliary shaft 16 can be varied continuously from -3000 to +3000 rpm, i.e. from 0 to 3000 rpm in either direction.

The input shaft 18 to the hydrostatic pump is  
25 additionally connected to a mechanical gearing unit generally designated 20. The gearing unit 20 is also connected to the auxiliary shaft 16 and has an output shaft 22. The latter output shaft is connected in a conventional manner through a differential to the driving wheels of the  
30 vehicle.

The gearing unit comprises two planetary gear sets that are generally designated 30 and 40, respectively. The first planetary gear set 30 is formed by a sun gear 32, planet  
35 gears 34 mounted on a carrier 36 and ring gear 38. Similarly, the second planetary gear set 40 is formed by a sun gear 42, planet gears 44 mounted on a carrier 46 and



ring gear 48. External teeth on ring gears 38 and 48 of both planetary gear sets are in permanent driving engagement with the output shaft 22 through gears 33 and 43 fixedly connected to the shaft 22.

5

The input shaft 18 carries a slidable coupling sleeve 35 that is slidable between different positions to couple the input shaft to one or both of the planetary gear sets 30 and 40, the sleeve 35 being shown in the position in which it is coupled to the first planetary gear set 30. Likewise, the auxiliary shaft 16 carries a slidable coupling sleeve 45 that is slidable between different positions to couple the auxiliary shaft to one or both of the planetary gear sets 30 and 40, the sleeve 45 also being shown in the position in which it is coupled to the first planetary gear set 30.

In the illustrated position of the coupling sleeves 35 and 45, the input shaft 18 is in driving engagement with the sun gear 32 of the first planetary gear set through pinions 50 and 54 and an idler 52. When the coupling sleeve 35 is moved upwards to its other end position, the input shaft 18 is instead connected to the carrier 46 of the second gear set by means of meshing pinions 56, 58. The sun gear 32 of the first planetary gear set 30 rotates in the same direction and with the same speed as the input shaft 18 whereas the carrier 46 of the second planetary gear set rotates at twice the speed in the opposite direction.

In the illustrated position of the sleeve 45, the auxiliary shaft 16 drives the carrier 36 of the first planetary gear set 30 through pinion 64. When the sleeve 45 is moved upwards to its other end position, the auxiliary shaft 16 is instead connected to the sun gear 42 of the second planetary gear set 40, through meshing pinions 60 and 62. The carrier 36 and that sun gear 42 rotate at the same speed as and in the opposite direction to the auxiliary shaft 16.

The two coupling sleeves 35 and 45 have an intermediate position in which drive is transmitted to the output shaft 22 from the input shaft 18 and the auxiliary shaft 16 through both planetary gear sets 30 and 40. This driving of the output shaft through the parallel torque paths of both planetary gear sets can only be allowed under specific conditions, to be described below, when both planetary gear sets drive the output shaft 22 at the same speed. Under all other condition, only one or other of the planetary gear sets 30 is engaged by moving the coupling sleeves upwards or downwards, as viewed. Downward movement of the sleeves 35 and 45 disengages the sleeves from the pinions 56 and 60 so that the second planetary gear set 40 is disengaged. Conversely, upward movement of the sleeves 35 and 45 uncouples the input and auxiliary shafts from the first planetary gear set 30.

Reference will now be made to Figure 2 to explain the operation of the transmission shown in Figure 1. Figure 2 is a diagram to enable the speed of rotation of any of the rotating elements to be calculated if the speed of the other two is known. In Figure 2, rotational speed is plotted along the vertical axis and the vertical lines S1, C1, R, C2 and S2 represent, respectively, speeds of the sun gear 32, carrier 36, the ring gears 38 and 48 (which have the same number of teeth and rotate at the same speed as one another), the carrier 46 and the sun gear 42. The separation of these vertical lines depends on the gearing ratios within the planetary gear sets. With the line representing the carrier arranged at unit distance from the line of the ring gear, the sun gear is represented by a line spaced from the line of the carrier by the ratio of the number of teeth in the ring gear to the number of teeth on the sun gear.

When rotational speeds plotted along any two of these vertical lines are connected to one another by a straight

line, the intersection of that straight line with the third vertical line will indicate the rotational speed of the remaining rotary element of the planetary gear set in question.

5

In Figure 2, the line R represents the speed of the two ring gears which are connected to the output shaft 22. The lines to the left of line R show how the output speed varies with the speed of the auxiliary shaft when the first  
10 planetary gear set 30 is engaged and those to the right of the vertical line R show how the output speed varies with the speed of the auxiliary shaft when the second planetary gear set 40 is engaged.

15 Considering first the operation of the first planetary gear set 30, here the sun gear is held at 2100 rpm (the speed of the input shaft 18) at all times. The speed of the first carrier 36 on the other hand can be varied from -3000 to +3000 rpm, as earlier assumed. From the lines drawn in  
20 Figure 2, it will be seen that the ring gears will rotate with a speed in the range from -3300rpm to +5000 rpm and will stop when the speed of the carrier is 600 rpm (all these values are approximate).

25 Considering next the operation of the second planetary gear set 40, here the carrier is driven at a constant speed of -4200 rpm (twice engine speed and in the opposite direction) while the sun gear 42 is driven by the auxiliary shaft 16 at a speed that can vary from +3000 to -3000 rpm.  
30 As a result, the ring gears are rotated at speeds from 5000 rpm to 9000 rpm in the direction of forward vehicle travel.

Thus in operation of the vehicle, the first gear set 30  
35 is engaged and the swash plate is moved from a first end position to a second to vary the speed of the vehicle from maximum reverse speed, through zero, to an intermediate

forward speed. To increase the forward speed to its maximum value, the second gear set 40 is engaged and the swash plate is back to its first end position

5        It will be noted that when the auxiliary shaft 16 is rotating at a speed of 3000 rpm then the ring gears are rotated by both planetary gear sets at the same speed of 5000 rpm. This overlap at the limit allows a power shift from one planetary gear set to the other, in that both  
10    planetary gear sets can safely be engaged temporarily so that no interruption in power transmission is required during the changeover.

      It will be appreciated that any number of gear ratios  
15    can be used in the planetary gear sets which will achieve a point of overlap when changeover can take place, more especially since the speed of the carrier 42 of the second planetary gear set can be set at will and need not be exactly double the engine speed. It is preferred however to  
20    set the gear ratios of the two planetary gear sets such that the changeover from one to the other occurs at one of the limits of adjustment of the swash plates of the hydrostatic pump and/or motor.

25        In the preferred embodiment of the invention, the sun gear 32 has twenty teeth and the ring gear 38 has fifty hence the line S1 being at a position 2.5 units away from the line C1. In the second planetary gear set 40, on the other hand, sun gear 42 has thirty four teeth and ring gear  
30    48 has fifty teeth so that the distance of the line S2 from the line C2 is just under 1.5 units.

      As it is important that the output shaft should be rotating at a predetermined speed when both planetary gear  
35    sets are engaged, it is preferred to provide a shaft speed sensor to initiate a changeover between gear sets instead of

relying on the swash plates being at the limits of their adjustment.

As previously mentioned, the invention can be applied  
5 to a hybrid electric vehicle. In such a vehicle, the  
hydraulic pump 10 and motor 14 are replaced by a generator  
charging a bank of storage batteries and an electric motor  
powered by the batteries. The engine can operate with  
constant speed when driving the input shaft 18 and the speed  
10 of the auxiliary shaft can be adjusted by control of the  
electric motor within the same limits as described above.

CLAIMS

1. A continuously variable transmission comprising  
an input shaft,  
5 a hydrostatic pump driven by the input shaft,  
a hydrostatic motor hydraulically connected to the  
hydrostatic pump and serving to drive an auxiliary shaft  
with a continuously variable transmission ratio relative to  
the input shaft, and  
10 mechanical gearing for transmitting torque from the  
input and auxiliary shafts to an output shaft,  
characterised in that the mechanical gearing comprises  
two planetary gear sets arranged to provide parallel torque  
paths from the input and auxiliary shafts to the output  
15 shaft and coupling means for selectively engaging the  
planetary gear sets,  
the first planetary gear set, when engaged by the  
coupling means to transmit torque, comprising first a sun  
gear connected to the input shaft, a first planetary gear  
20 carrier connected to the auxiliary shaft and a first ring  
gear connected to drive the output shaft,  
the second planetary gear set, when engaged by the  
coupling means to transmit torque, comprising a second sun  
gear connected to the auxiliary shaft, a second planetary  
25 gear carrier connected to the input shaft and a second ring  
gear connected to drive the output shaft, and  
the coupling means serving to engage only the first  
planetary gear set for reverse and lower transmission  
ratios, to engage only the second planetary gear set for  
30 higher transmission ratios and to engage both planetary gear  
sets simultaneously when the ratio of the speed of rotation  
of the auxiliary shaft relative to the input shaft is such  
that both planetary gear sets drive the output shaft at the  
same speed.  
35
2. A transmission as claimed in claim 1, wherein the  
transmission ratios of the planetary gear sets are selected

such that the planetary gear sets drive the output shaft at the same speed only at the limit of adjustment of the speed of the auxiliary shaft relative to the input shaft.

5           3.    A transmission as claimed in claim 1 or 2, wherein, in the first planetary gear set, the number of teeth on the ring gear are substantially in the ratio of 5:2 to the number of teeth on the sun gear.

10           4.    A transmission as claimed in any preceding claim, wherein, in the second planetary gear set, the number of teeth on the ring gear are substantially in the ratio of 3:2 to the number of teeth on the sun gear.

15           5.    A continuously variable transmission of a hybrid electrical vehicle comprising:

          an input shaft connected to an internal combustion engine,

          an auxiliary shaft driven by an electric motor  
20    deriving power from a bank of batteries charged by the internal combustion engine, the auxiliary shaft being rotatable by the electric motor with a speed that is continuously variable relative to the speed of rotation of the input shaft, and

25           mechanical gearing for transmitting torque from the input and auxiliary shafts to an output shaft,

          characterised in that the mechanical gearing comprises two planetary gear sets arranged to provide parallel torque paths from the input and auxiliary shafts to the output  
30    shaft and coupling means for selectively engaging the planetary gear sets,

          the first planetary gear set, when engaged by the coupling means to transmit torque, comprising first a sun gear connected to the input shaft, a first planetary gear  
35    carrier connected to the auxiliary shaft and a first ring gear connected to drive the output shaft,

the second planetary gear set, when engaged by the coupling means to transmit torque, comprising a second sun gear connected to the auxiliary shaft, a second planetary gear carrier connected to the input shaft and a second ring gear connected to drive the output shaft, and

the coupling means serving to engage only the first planetary gear set for reverse and lower transmission ratios, to engage only the second planetary gear set for higher transmission ratios and to engage both planetary gear sets simultaneously when the ratio of the speed of rotation of the auxiliary shaft relative to the input shaft is such that both planetary gear sets drive the output shaft at the same speed.

6. A transmission as claimed in claim 5, wherein the transmission ratios of the planetary gear sets are selected such that the planetary gear sets drive the output shaft at the same speed only at the limit of adjustment of the speed of the electric motor relative to the input shaft.

7. A transmission as claimed in claim 5 or 6, wherein, in the first planetary gear set, the number of teeth on the ring gear are substantially in the ratio of 5:2 to the number of teeth on the sun gear.

8. A transmission as claimed in claim 5, 6 or 7, wherein, in the second planetary gear set, the number of teeth on the ring gear are substantially in the ratio of 3:2 to the number of teeth on the sun gear.

9. A transmission as claimed in any preceding claim, comprising a shaft speed sensor associated with the output shaft for permitting torque to be transmitted through both planetary gear sets only when the speed of output shaft is at a predetermined value.



10. A continuously variable transmission constructed, arranged and adapted to operate substantially as herein described with reference to and as illustrated in the accompanying drawings.



Application No: GB 0205213.2  
Claims searched: 1-10

Examiner: Joe Mitchell  
Date of search: 2 July 2002

**Patents Act 1977**  
**Search Report under Section 17**

**Databases searched:**

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:  
UK Cl (Ed.T): F2D (DEA, DEB, DEC, DED)  
Int Cl (Ed.7): F16H 3/72, 37/08  
Other: Online: WPI, EPODOC, JAPIO.

**Documents considered to be relevant:**

Category	Identity of document and relevant passage	Relevant to claims
A	GB 1593750 A THYSEN INDUSTRIE AG	
A	US 6260440 B1 CRONIN et al.	
A	US 5667452 A COUTANT	
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A	US 3869939 A MIYAO et al.	

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